PARAMETRIC THERMODYNAMIC ANALYSIS OF INTERCOOLED AND INTERCOOLED-RECUPERATED GAS TURBINE BASED CYCLES

[1]MITHILESH KUMAR SAHU, [1]TUSHAR CHAUDHARY, [1]ANUPAM KUMARI, [2]SANJAY

[1] Ph.D. Scholar, Mechanical Engg. Deptt. NIT Jamshedpur;

[2] Professor, Mechanical Engg. Department, National Institute of Technology, Jamshedpur, INDIA;

E-mail: mithleshsahu3646@gmail.com

ABSTRACT: - Gas turbine based power plants are considered as most promising and efficient source of power generation. In this paper, focus is on the performance evaluation of intercooled gas turbine (IcGT) and intercooled-recuperated gas turbine (IcRcGT) cycle. Performance of a gas turbine is mainly affected by the various parameters like pressure ratio, turbine inlet temperature and ambient conditions. The evaluation is done on the basis of thermodynamic analysis, computed with the help of energy and exergy analysis. The analysis of various cycle components is done by adopting the energy and exergy balance approach. When operating at high turbine-inlet-temperature (TIT) turbine blade cooling is required so the cooled gas turbine has been analyzed where turbine blades are cooled by compressor bled air. Blade cooling increases the life of blade and reduces gas turbine work output due to mixing losses. The recuperation technique has also been integrated to the cycle for utilization of high temperature turbine exhaust. The energy efficiencies found for IcGT cycle is 35.07% and for IcRcGT cycle it is 43.1% respectively.

1. INTRODUCTION

Gas turbine burning natural gas is a promising energy conversion system for power generation due to its lower emission characteristics as the energy it provides is clean. 21st century requires such a source of power generation which fulfills the need of power and provides energy security along with lower emission. Its high energy conversion efficiency and large output with reduced emission makes it the need of future power demand [1]. Exergy analysis has been integrated with the energy analysis as a useful tool to determine thermodynamic inefficiencies in components of the thermodynamic cycles. In this field of research H. Chandra et.al. have reported the energy and exergy analysis of simple closed Brayton thermal power cycle [2] and have plotted results showing energy and exergy destroyed in various components of the cycle. Adoption of higher turbine inlet temperatures in an effort to improve thermodynamic energy conversion efficiency requires the adoption of advanced blade cooling methods. Blade cooling allows for high inlet temperature of turbine but also causes a decrease in net power output [3]. The reduction in net work is overcome by implementing the intercooling. The use of intercooler reduces the power requirement of compressor, the reduction in compression work results in a corresponding increase in output of the gas turbine [4]. The efficiency of gas-turbine cycles can be enhanced by the use of auxiliary equipment such as intercoolers, regenerators, and reheaters. These devices are bulky hence expensive, however, and economic considerations usually preclude their use. Intercooling and recuperation are known to be the means to improve specific output and efficiency respectively of a gas turbine. Colin F. Mc Donald et. al. [5] has dealt with the utilization of recuperated and regenerated engine cycle for high efficiency gas turbine in 21st century. R. Bhargava et. al [6] reported the approach to increase the efficiency with intercooling and cogeneration process which leads the integration of this process with the simple Brayton cycle for higher degree of performance. The energy and exergy analysis of steam cooled reheat gas steam combined cycle [7] reports that using closed loop steam cooling the plant thermal efficiency can be improved up to 62%. The effect of turbine blade cooling on cycles [8], defining the cooled turbine efficiency [9], and J.H. Horlock [10] worked on amount of blade coolant air required. Sanjay et. al. [11] worked on comparative performance analysis of cogeneration gas turbine cycle for different blade cooling means. Considering the above literature and by applying energy and exergy balance approach proposed cycles have been analyzed.

NOMENCLATURE

c_p	specific heat at constant pressure $(\frac{kJ}{kg\cdot K})$
$E_{X,H}$	total exergy supplied (kW)
ex	specific exergy of the stream $(\frac{kJ}{kg})$
h	specific enthalpy of the stream $(\frac{kJ}{kg})$
$\dot{m}$	mass flow rate $(\frac{kg}{s})$
p	pressure (bar)
p_o	reference or ambient pressure (kPa)
T_0	reference or ambient temperature
Q	heat transfer rate (kW)
Q_H	heat supplied by fuel (kW)
q_H	heat gain by working fluid (kW)
s	specific entropy $(\frac{kJ}{kg\cdot K})$
S	entropy $(\frac{kJ}{K})$
S_{gen}	entropy generation $(\frac{kJ}{K})$
S(stream)	represents energy content in a stream computed by the product of mass flow rate and enthalpy of that stream (kW)
T	temperature (K)
T_o	reference or ambient temperature (K)
W	work (kW)

SUBSCRIPTS

a	air/ambient
c	compressor
comb	combustion chamber
ex	exergy
d	destruction
Gen	generation
f	fuel
G	gas
in	inlet
Out	outlet
Ic	intercooler
sat	saturated
T	turbine
w	water
I	first law

II	second law
1, 2, 3…	state points

GREEK SYMBOLS

$\varnothing$	thermodynamic function
ε	effectiveness
η	efficiency
ω	availability per unit mass of gas
Ω_d	exergy destruction rate

ACRONYMS

A	alternator
BGT	brayton gas turbine cycle
C	Compressor
CMBEXT	combustion exit
FRMRCP	from recuperator
GTCOMB	gas turbine combustion chamber
HPT	high pressure turbine
HPC	high pressure compressor
HP NOZZ	high pressure turbine nozzle
HP ROTR	high pressure turbine rotor
I	irreversibility
IPT	intermediate pressure turbine
IC	intercooler
IcGT	intercooled gas turbine
IcRcGT	intercooled recuperated gas turbine
IPNOZZ	intermediate pressure turbine nozzle
IPROTR	intermediate pressure turbine rotor
LPC	low pressure compressor
PT	power turbine
RECUP	recuperator
R_BYPS	recuperator bypass
TORCUP	to recuperator
WATRES	water reservoir

2. SYSTEM DESCRIPTION

Fig.1 shows schematic diagram of intercooled gas turbine (IcGT) cycle. The key features of this proposed cycle is water cooled intercooler in between two stages of compressors. Also the expansion stages of gas turbine incorporate compressor bled air cooling. The coolant air is bled from the high pressure compressor (HPC) at an appropriate pressure level and is allowed to enter the hollow gas turbine blades from the root. The coolant air is passing through a serpentine path before exiting the blade at its tip and mixing with the main flow gases. Other aspects of cycle are similar to conventional gas turbine cycle.

Fig.2 shows a schematic diagram of an intercooled-recuperated gas turbine cycle. This cycle differs from the previously discussed proposed cycle (IcGT) as it incorporates recuperator which successfully extracts a significant amount of thermal energy exiting the power turbine (PT). This feature is achieved by the help of a splitter unit, which divides the HPC exit stream to the recuperater unit and MIXRCP unit.

3. MODELING AND GOVERNING EQUATIONS

3.1 Gas model

The inlet ambient air has been assumed to be at 1 bar and 288K with relative humidity of 50% and natural gas as a fuel. Gas model is based on the assumption that specific heat of gas is a function of temperature at constant pressure and is given by the polynomial:

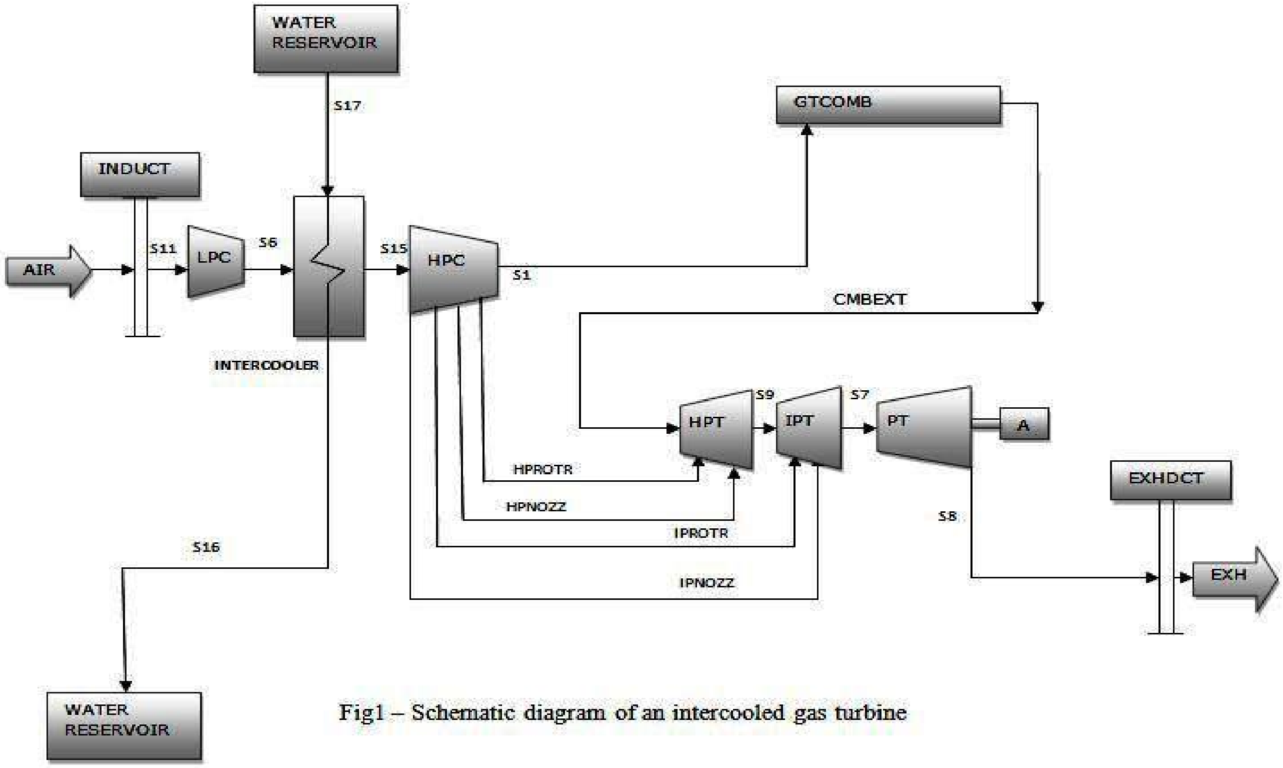

Fig1 – Schematic diagram of an intercooled gas turbine

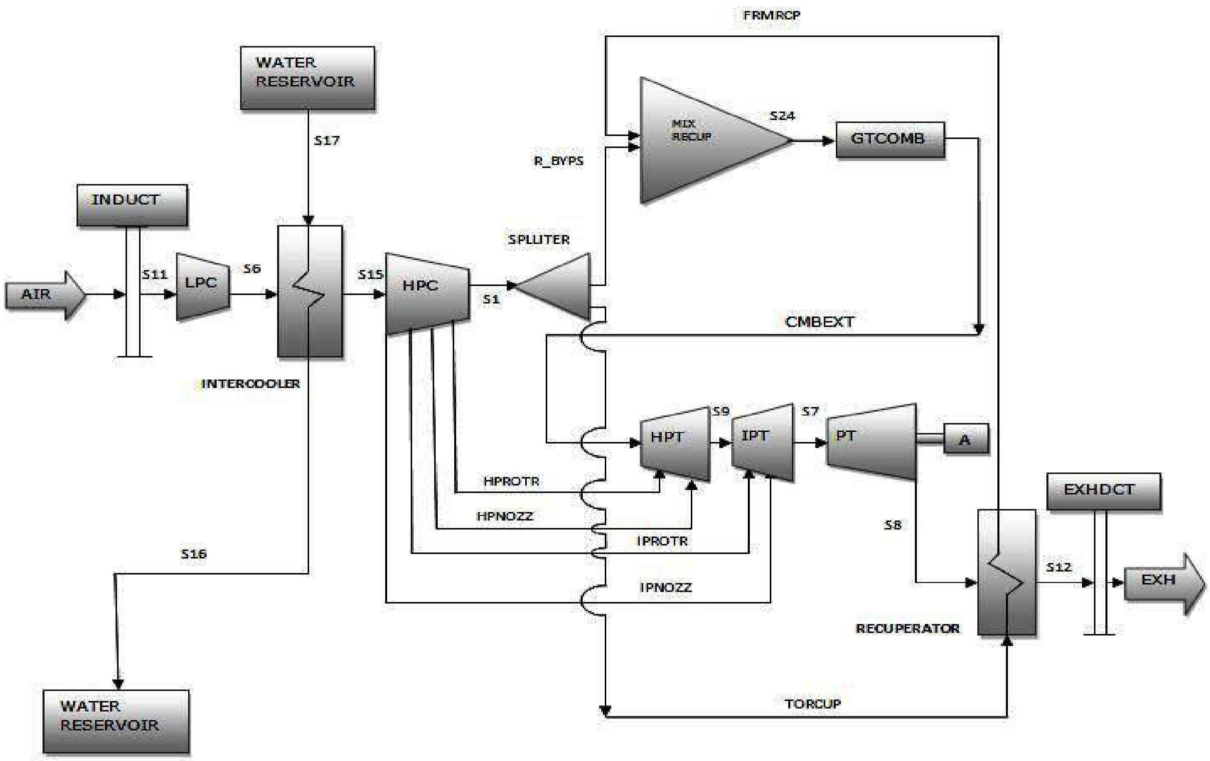

Fig2. – Schematic diagram of an intercooled recuperated gas turbine

3. MODELING AND GOVERNING EQUATIONS

3.1 Gas model

The inlet ambient air has been assumed to be at 1 bar and 288K with relative humidity of 50% and natural gas as a fuel. Gas model is based on the assumption that specific heat of gas is a function of temperature at constant pressure and is given by the polynomial:

$$c_p(T) = a + bT + cT^2 + dT^3 + \ldots\ldots\ldots\ldots \quad (1)$$

Where a, b, c, d, etc. are the coefficients of the polynomial, and their values are taken from the work of Toulounkain and Tadash [12].
From above polynomial enthalpy, entropy and exergy has been calculated with the help of following equations:

$$h = \int_{T_a}^{T} c_p(T)dT \quad \ldots\ldots\ldots\ldots\ldots \quad (2)$$

$$\Phi = \int_{T_a}^{T} c_p(T).\frac{dT}{T} \quad \ldots\ldots\ldots\ldots\ldots \quad (3)$$

$$s = \Phi - R\ln\left(\frac{p}{p_a}\right) \quad \ldots\ldots\ldots\ldots\ldots \quad (4)$$

$$\omega = h - T_\circ \cdot s = h - T_\circ.\Phi + R.T_\circ.\ln\left(\frac{p}{p_\circ}\right) \ldots\ldots \quad (5)$$

Here, all non-reacting gases are assigned zero thermodynamic enthalpy, entropy and availability at the ambient conditions.

3.2 Compressor model

Compression is assumed to be Polytropic. The governing equations related to thermodynamic performance of compressor is as under:

From energy analysis

Work input to the compressor (W_c) = Energy content at compressor outlet stream – Energy content at compressor inlet stream ………… ……… (6)

$$W_{LPC} = m_6 \cdot h_6 - m_{11} \cdot h_{11} \quad \ldots\ldots\ldots\ldots \quad (7)$$

First law thermodynamic efficiency,

$$\eta_{I,c} = \frac{\text{work required to the compressor}}{\text{actual work supplied}} \quad \ldots\ldots\ldots \quad (8)$$

$$W_C = W_{LPC} + W_{HPC} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (9)$$

From exergy analysis

$$I_c = \sum \dot{m}_{in} ex_{in} - \sum \dot{m}_{out} ex_{out} + W_c \quad \ldots\ldots\ldots \quad (10)$$

which gives,

$$T_{\circ}\Delta S_{gen,c} = \dot{m}T_{\circ}(s_{in} - s_{out}) \qquad \text{............} \qquad (11)$$

The second law efficiency of compressor is as under:

$$\eta_{II,c} = 1 - \frac{I_c}{W_c} \qquad \text{.................} \qquad (12)$$

3.3 Intercooler model

The increase in cycle net-work output may be achieved by the adoption of multi-stage intercooled compressor layout since inter-cooler saves some work of compression. The intercooler may be surface or evaporative type, with surface type being more common. Practically an inter-cooler effectiveness is always less than 100% and thus Intercooling is always imperfect and pressure losses occur in both the fluid streams. The following assumptions have been made for the mathematical modeling of intercooler:

- Cooling medium, water is taken at ambient conditions.
- Intercooler selected for the analysis is of surface type.
- Intercooler is a counter flow liquid-air type heat exchanger
- Losses in intercooler are accounted by assuming appropriate values of intercooler-effectiveness (∈) and pressure drop (Δp) in the air-stream side as detailed in Table 1.

The effectiveness of intercooler is given by the following equation:

$$\varepsilon_{ic} = \frac{(T_{ic,a})_{in} - (T_{ic,a})_{out}}{(T_{ic,a})_{in} - (T_{ic,w})_{in}} \qquad \text{...............} \qquad (13)$$

Energy balance of intercooler gives:

$$\dot{m}_{ic,a}.c_{p,a}.\varepsilon_{ic}.((T_{ic,a})_{in} - (T_{ic,a})_{out}) - \dot{m}_{ic,w}.c_{p,w}.\{(T_{ic,w})_{out} - (T_{ic,w})_{in}\} = 0 \qquad \text{........} \qquad (14)$$

Exergy destroyed is given as:

$$(\Omega_{ic})_d = (\dot{m}_{ic,a})_{in}.\ \{(\omega_{ic,a})_{in} - (\omega_{ic,a})_{out}\} + \dot{m}_{ic,w}\{(\omega_{ic,w})_{in} - (\omega_{ic,w})_{out}\} \qquad \text{......} \qquad (15)$$

$$I_{ic} = \sum \dot{m}_{in}ex_{in} - \sum \dot{m}_{out}ex_{out} \qquad \text{....} \qquad (16)$$

3.4 Combustion chamber model

The combustion process is accompanied by certain losses including incomplete combustion and loss of pressure. The fuel flow has been determined from the mass and energy balance, and the exergy destroyed has been obtained from the exergy balance.

$$\dot{m}_{comb} = \dot{m}_a + \dot{m}_f \qquad \text{...............} \qquad (17)$$

Combustion chamber in-efficiency leads to energy and exergy loss within the component.
Energy and exergy balance method has been used to determine the energy and exergy loss in the GTCOMB as under:
Energy loss = (Energy content at outlet stream) – (Energy content at inlet stream)

$$\text{Energy loss} = \dot{m}_{CMBEXT}.h_{CMBEXT} - \dot{m}_{24}.h_{24} - \dot{m}_f.CV \qquad \text{......} \qquad (18)$$

$$\text{Percentage energy loss} = \frac{\text{Energy loss}}{\text{Energy supplied}} \qquad \text{...} \qquad (19)$$

Exergy balance is given as:

$$0 = q_H(1-\frac{T_\circ}{T_s}) + \dot{m}(ex_{in} - ex_{out}) - T_\circ \Delta S_{gen,c} \quad \ldots \tag{20}$$

$$I_{comb} = \sum \dot{m}_{in} ex_{in} - \sum \dot{m}_{out} ex_{out} + \dot{m}_f.CV.\left(1-\frac{T_\circ}{T_H}\right) \quad \ldots. \tag{21}$$

which gives,

$$\text{Exergy destroyed due to irreversibility} = \dot{m} T_\circ \Delta S_{gen,c} \tag{22}$$

The second law efficiency,

$$\eta_{II,comb} = 1 - \frac{I_{comb}}{E_{X,H}} \quad \ldots\ldots \tag{23}$$

3.5 Cooled Gas Turbine model

Gas turbine operating at high temperature needs cooling for safe operation. The purpose of the blade cooling is to keep the blade temperature to a safe level, and blade cooling ensures a long creep life, low oxidation rates, and low thermal stresses.
The blade coolant requirement model has been adopted from author's previous work [4]:

$$\frac{\dot{m}_c}{\dot{m}_g} = \bar{St}_{in} \cdot \left(\frac{S_g}{t\cos\alpha}\right) \cdot F_{sa} \cdot \left(\frac{c_{p,g}}{c_{p,c}}\right) \cdot \left(\frac{T_{g,i} - T_b}{\varepsilon \cdot (T_b - T_{c,i})}\right) \tag{24}$$

The expansion process in the turbine has been modeled as under:
First law efficiency of gas turbine is given as under:

$$\eta_I = \frac{W_T}{q_H} \quad \ldots\ldots\ldots. \tag{25}$$

Exergy balance equation for turbine,

$$W_T = \dot{m}\left(ex_{in} - ex_{out}\right) - T_o \Delta S_{gen} \quad \ldots\ldots \tag{26}$$

$$I_T = \sum \dot{m}_{in} ex_{in} - \sum \dot{m}_{out} ex_{out} - W_T \quad \ldots \tag{27}$$

Which gives,

$$T_o \Delta S_{gen} = \dot{m} T_o (s_{out} - s_{in}) \quad \ldots. \tag{28}$$

$$\eta_{II,T} = 1 - \frac{I_T}{\sum \dot{m}_{in} ex_{in} - \sum \dot{m}_{out} ex_{out}} \quad \ldots\ldots.. \tag{29}$$

4. Result and Discussion: -

Calculations made by using the detailed energy and exergy balance equation gives the energy loss/utilized in different components of cycle and the exergy destruction of the cycle. Fig. 1 and Fig. 2 shows the cycle configuration of the IcGT and IcRcGT respectively, which explain to and from motion of the streams. Table 1 and 2 show the state points of the cycle in terms of its

Table 1: Operating parameters of IcGT cycle

STREAM	FROM	TO	P (bar)	T (K)	ṁ (kg/s)	h (kJ/kg)	ex (kJ/kg)
SEA-IN	-	WATRES	1.72	288.55	88.19	2.1	70.47
S6	IPC	INTCOL	3.19	415.89	62.99	129.09	116.74
S7	IPT	PT	5.06	1116.3	64.45	936.51	610.99
S8	PT	EXHDCT	1.04	790.92	64.45	555.11	238.12
s9	HPT	IPT	7.67	1288.2	58.61	1152.1	798.49
s11	INDUCT	IPC	0.99	287.99	62.99	0	0
S15	INTCOL	HPC	3.06	305.1	62.99	17.18	92
S16	INTCOL	WATRES	3.64	320.32	58.25	135.53	177.55
S17	WATRES	INTCOL	3.79	294.97	58.25	29.3	176.09
S1	HPC	GTCOMB	15.33	505.39	47.06	224.73	284.12
CMBEXT	GTCOMB	HPT	14.52	1644.1	48.52	1615.9	1207.1
AIR(dead)	-	INDUCT	1.01	288	62.99	0	0
HPNOZZ	HPC	HPT	15.33	505.39	7.68	220.65	280.04
IPROTR	HPC	IPT	8.96	427.89	1.16	141.3	205.9
HPROTR	HPC	HPT	15.33	505.39	2.41	220.65	280.04
IPNOZZ	HPC	IPT	8.96	427.89	4.66	141.3	205.9

Table 2: Operating parameters of IcRcGT cycle

STREAM	FROM	TO	P (bar)	T(K)	ṁ(kg/s)	h (kJ/kg)	ex(kJ/kg)
S6	IPC	INTCOL	3.19	415.89	62.99	129.09	116.74
S7	IPT	PT	5.23	1129.3	64.18	952.96	620.02
S8	PT	RECUP	1.04	790.92	64.18	555.11	238.1
s9	HPT	IPT	7.94	1302.3	58.34	1171.2	812.26
s11	INDUCT	IPC	0.99	287.99	62.99	0	0
EXH	EXHDCT	-	1.01	619.11	64.18	359.15	120.69
S12	RECUP	EXHDCT	1.02	619.11	64.18	359.15	121.31
S15	INTCOL	HPC	3.06	305.1	62.99	17.18	92
S16	INTCOL	WATRES	3.64	320.32	58.25	135.53	177.55
S17	WATRES	INTCOL	3.79	294.97	58.25	29.3	176.09
S1	HPC	COLSPL	15.33	505.39	47.06	224.73	284.12
SEAOUT	WATRES	-	1.72	307.8	88.19	83.7	74.27
CMBEXT	GTCOMB	HPT	14.52	1644.1	48.25	1615.9	1207.1
S24	MIXRCP	GTCOMB	14.87	753.63	47.06	479	342.66
AIR(dead)	-	INDUCT	1.01	288	62.99	0	0
R_BYPS	COLSPL	MIXRCP	15.33	505.39	2.35	220.65	280.04
HPNOZZ	HPC	HPT	15.33	505.39	7.68	220.65	280.04
IPROTR	HPC	IPT	8.96	427.89	1.16	141.3	205.9
HPROTR	HPC	HPT	15.33	505.39	2.41	220.65	280.04
IPNOZZ	HPC	IPT	8.96	427.89	4.66	141.3	205.9
TORCUP	COLSPL	RECUP	15.33	505.39	44.75	220.65	280.04
FRMRCP	RECUP	MIXRCP	14.87	766.35	44.71	492.87	427.37

temperature, pressure, flow rate, enthalpy and exergy. Table 3 relate to the performance parameters used for the analysis of cycle components.

Table 3: Performance parameters used for analysis of the proposed cycles.

Parameter	Symbol	Unit
Gas properties	$c_p = f(T)$	kJ kg^{-1} k^{-1}
	Enthalpy h $= \int c_p(T)\,dT$	kJ kg^{-1}
Compressor	Polytropic efficiency = 92.0	%
	Mechanical efficiency = 98.5	%
Combustor	Combustor efficiency = 99.5	%
	Air fuel ratio = 43.44 (IcGT) Air fuel ratio = 53.12 (IcRcGT)	-
Gas turbine	Polytropic efficiency = 92.0	%
	Exhaust pressure = 1.04	bar
Intercooler	Effectiveness =0.9162	-
Recuperator	Effectiveness =0.6017	-
Mass flow rate of water	58.258	kg/s
Mass flow rate of air	62.9988	kg/s

Fig.3 is a pie chart which shows the energy distribution within the IcGT cycle. Fig.3 helps us to understand that the exhaust loss is 52.04% and the overall efficiency of cycle is 35.07%. The loss takes place in compressor; turbine and combustion chamber are 0.35%, 1.34% and 0.5% respectively.

Fig.4 shows the energy balance of the IcRcGT the fig shows that exhaust stream takes away the 41.4% of total energy supplied and provides the cycle efficiency of 43.1%. The remaining part of energy is lost in other components like compressor, turbine and combustion chamber.

Fig.5 shows the percentage of exergy destruction of different components within the IcGT cycle. To measure actual losses and to perform the accurate thermodynamic analysis, exergy analysis is advantageous. Exergy analysis provides the actual performance parameters of the cycle. In pie chart it can be clearly seen that the maximum amount of exergy supplied is destroyed in exhaust which amounts 27.3% of total exergy supplied and followed by the combustion chamber 19.5%. The sum of percentage of exergy destruction in compressor, turbine and intercooler amounts 10.5%.

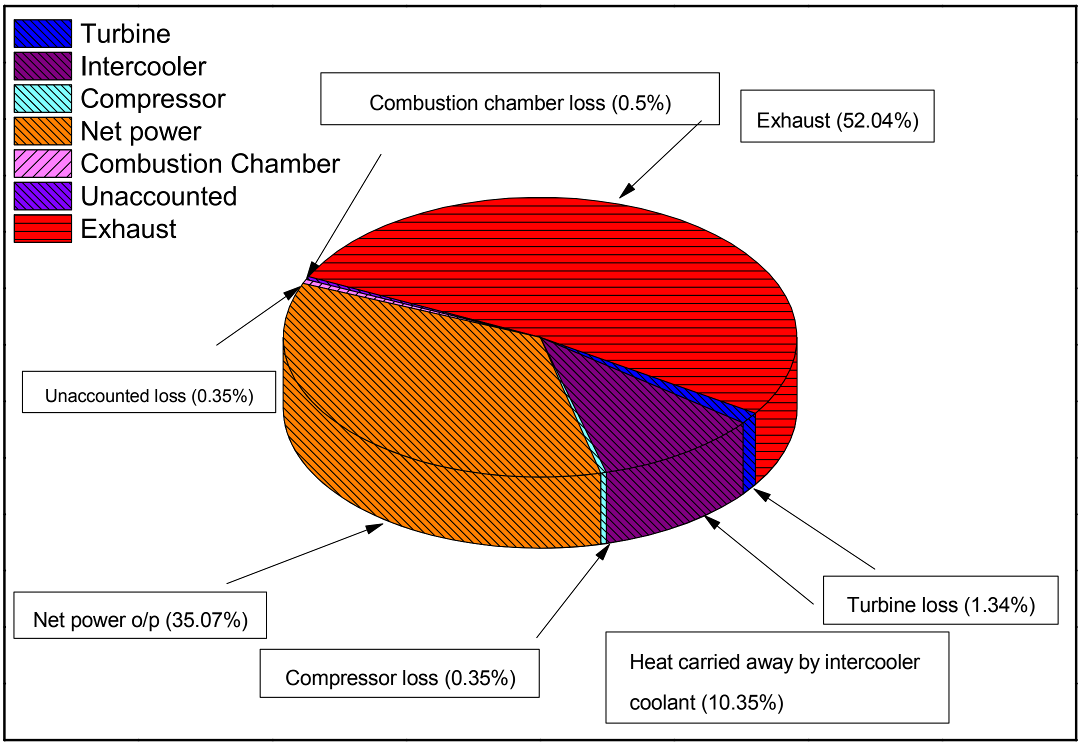

Fig. 3: Percentage energy balance in intercooled gas turbine

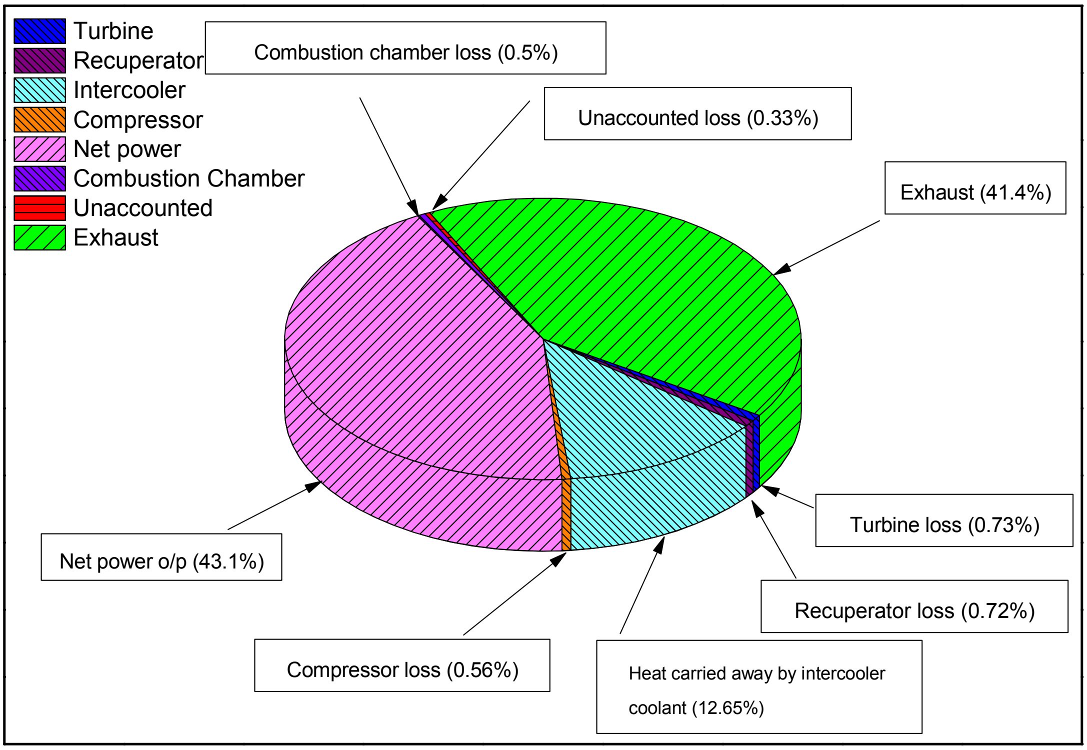

Fig. 4: Percentage energy balance in intercooled recuperated gas turbine

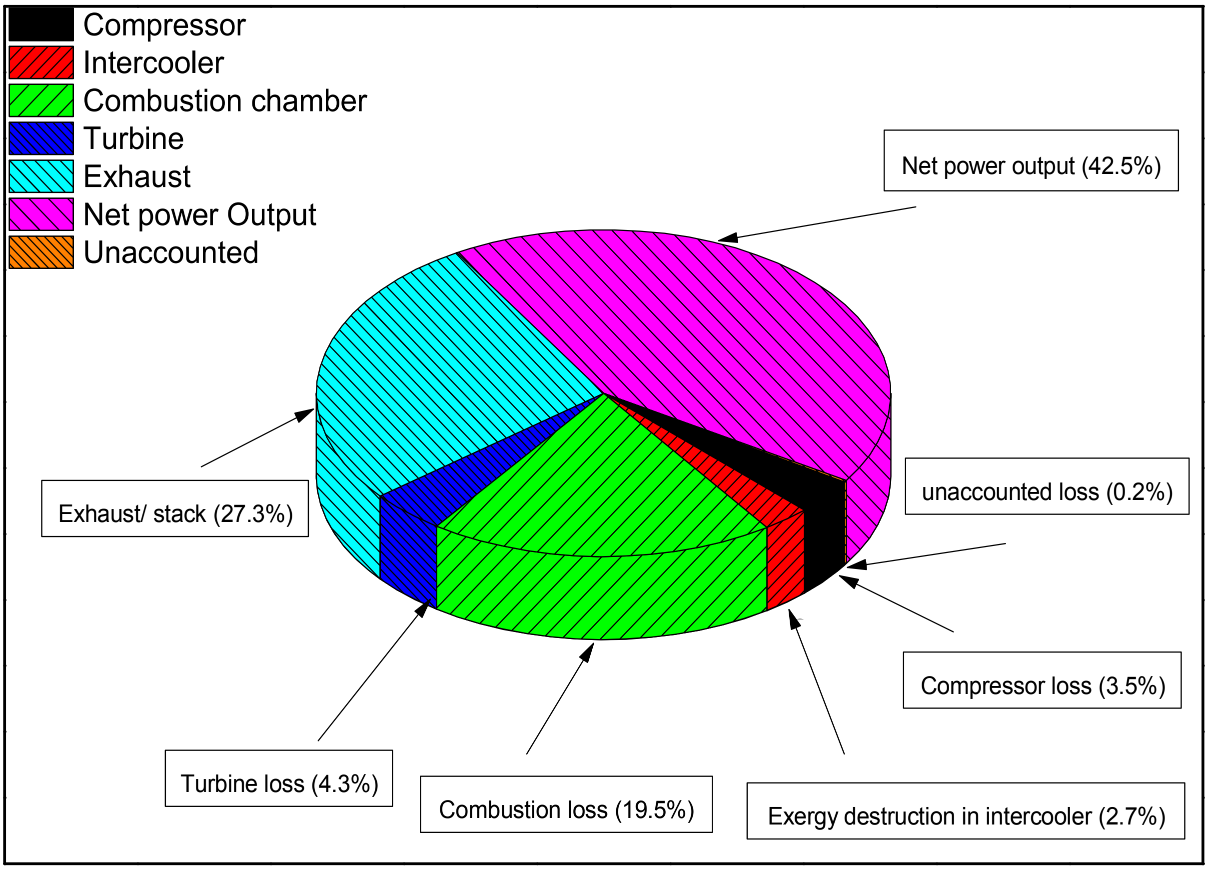

Fig. 5: Component-wise percentage exergy destruction in intercooled gas turbine cycle.

Fig.6 depicts the exergy destruction of different component of IcRcGT cycle. From the chart it is clear that maximum amount of exergy is destroyed in exhaust gas stream 16.95% followed by that in the combustion chamber 8.3%. The summation of exergy destruction occurring in remaining cycle component turbine, compressor, recuperator, and intercooler amounts to 17.14 %.

Fig.7 shows comparative data related to performance of intercooled gas turbine cycle with intercooled-recuperated gas turbine [8]. It shows that exergy efficiencies for intercooled gas turbine cycle (IcGT) is 42.5% and while that in IcRcGT cycle it enhances and is 52.25%. The exergy destruction associated with major components of both cycles (IcGT and IcRcGT) have been plotted. The graph shows that the exergy destruction in compressor for IcGT is 3.5% whereas for IcRcGT it is about 4.26%. The exergy destruction in gas turbine is around 4.3% in IcGT and 7.71% for IcRcGT. The exhaust gas loss in IcGT is higher as compared to IcRcGT, the reason being the absence of recuperator which partially utilizes heat content of exhaust gas stream. The sum total of exergy destruction in major cycle components (GTCOMB, GT, C and stack) is 37.27% for IcRcGT and 54.3% for IcGT.

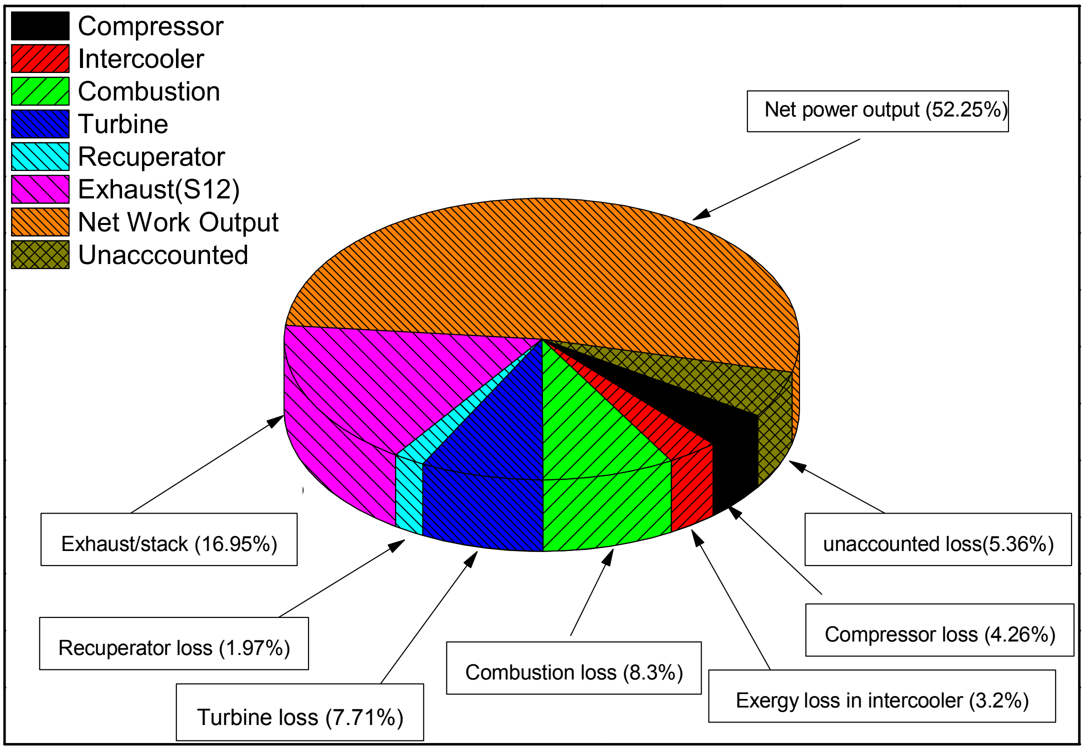

Fig. 6: Component-wise percentage exergy destruction in intercooled recuperated gas turbine cycle.

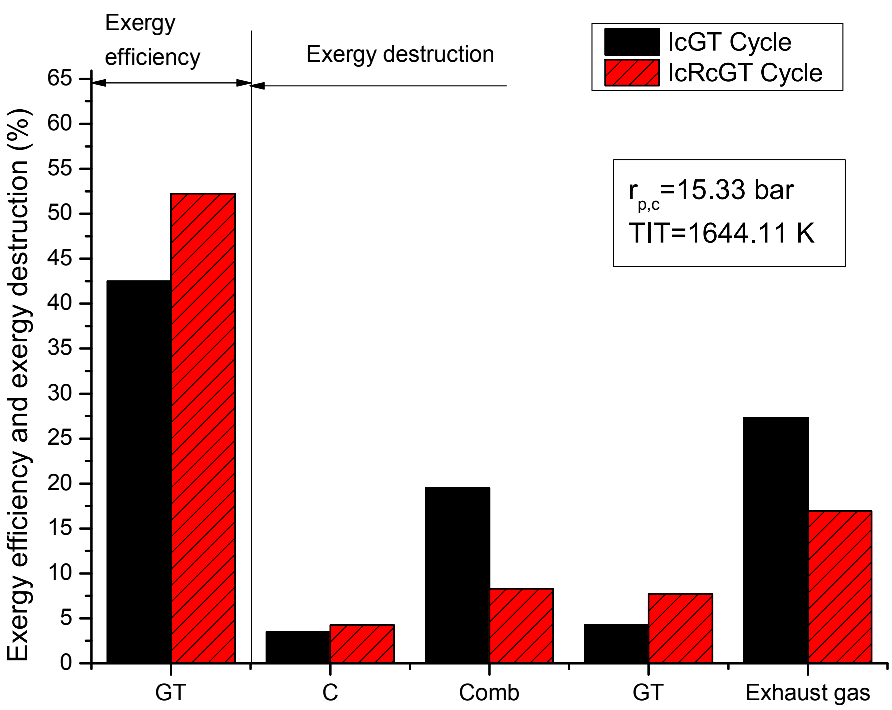

Fig 7: Effect of cycle configuration on exergy efficiency and component wise exergy destruction.

5. CONCLUSION

Based on the comprehensive thermodynamic analysis of complex intercooled recuperated gas turbine cycle, the following conclusions have been drawn:

1. Component level thermodynamic analysis suggests that losses arise due to irreversibility's within the components of the cycle.
2. The increase in energy efficiency and exergy efficiency has been observed in comparison to the intercooled gas turbine cycle.
3. The Pressure loss is also taking into consideration and seen that the pressure loss is highest in the recuperator and is around 0.46 bar followed by the combustion chamber around 0.35 bar and followed by that in intercooler in IcRcGT cycle.
4. Exergy destruction at component level is maximum in exhaust gas stream at 16.95% followed by combustion chamber at 8.30% in IcRcGT whereas in IcGT cycle it is 27.3% and 19.5% respectively.
5. Thermodynamic analysis of complex gas turbine cycle shows that a considerable amount of exergy is destroyed in exhaust that is a major loss of exergy supplied.
6. The utilization of turbine exhaust gas by using recuperator improves the cycle efficiency which can be seen in the fig 7 which shows the exergy efficiencies of the complex gas turbine cycles

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